Numerical Modeling on Reciprocating Active Magnetic Refrigeration Regenerator at Room Temperature

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ABSTRACT

A 2-dimension porous media model has been constructed for Reciprocating Active Magnetic Refrigeration Regenerator (AMR). The 2-D model is solved using a fully implicit (in time and space) discretization of the 2-dimension Navier-Stocks equations and governing energy equation. The magneto-caloric effect (MCE) is taken into account by the inclusion of a source term in the energy equation for the magnetic solid. The solid magnetic material and the regeneration fluid are separately modeled. The adiabatic temperature change of the gadolinium is used for mean field modeling. Typical results are presented based on the condition of experimental AMR.

INTRODUCTION

Magnetic refrigeration makes use of the magneto-caloric effect (MCE) which will be intrinsic to most magnetic materials, and therefore, a lot of cooling applications based on the MCE have been considered. One of the advantages of magnetic refrigeration is that the variation of temperature can be changed by an adiabatic magnet field change. Magnetic refrigeration has been used in cryogenic refrigeration since the 1930's. Operation near room temperature recently became feasible with the increased interest in alternative cooling techniques and the availability of large MCE materials and high field permanent magnets. Today, one of the most common designs for magnetic refrigeration is the AMR which makes possible the wide temperature spans with high efficiency and high cooling power.

AMR is also an environmentally attractive alternative to vapor compression refrigeration because it does not use a fluorocarbon working fluid and has the potential to be more efficient. Moreover, the magnetic refrigeration system may be possible to be compact because of the higher entropy density of magnetic material compared with that of a refrigerant gas.

In order to fully understand the physical phenomenon and predict the performance of AMR, a reliable numerical model is required. Until now, the 1-dimension numerical model based on the determination of the convective heat transfer coefficient between solid and fluid and the friction factor to specify the pressure drop in the fluid due to viscosity is used by many investigators such as Allab et al.[1] The results generated by the 1-D model, although valuable, show discrepancies with experimental data, but the differences are mainly caused by the use of the heat transfer correlation.

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To eliminate the use of heat transfer and pressure drop correlations, the velocity, pressure and temperature field must to be solved simultaneously. In doing so, a 2-dimension porous media model close to reality has been modeled.

The 2-D model is based on the two-dimensional Navier-Stokes equations for the fluid flow around the discrete particles which are alternatively heated and cooled during magnetization and demagnetization. The adopted numerical solution is explained and typical results are presented and analyzed.

AMR CYCLE

The AMR cycle is needed to realize a continue cooling. The magnetic Brayton cycle is the most basic one used in AMR cycle, which consists of two adiabatic processes and two constant magnetic field processes. The four steps in this cycle are described as follows:

1. Adiabatic magnetization: The magnetic material is placed in an adiabatic condition, and the heat transfer fluid doesn't flow. After magnetization, the temperature is up to $(T + \Delta T_{ad})$ due to the MCE.
2. Cold to Hot flow at $B=B_{max.}$ The fluid flows from the cold end to the hot end, rejecting heat to the hot reservoir. Then, the temperature of magnetic material is back to T.
3. Adiabatic demagnetization. The magnetic material is in another adiabatic condition where the magnet field changes from $B_{max} > 0$ to $B_{min}=0$, and the fluid doesn't flow. After demagnetization, the temperature is down to $(T - \Delta T_{ad})$.
4. Hot to cold flow at $B=B_{min}$. Pushing the fluid back from hot end to cold end, makes the temperature back to T, absorbing heat from cold reservoir.

NUMERICAL METHOD OF 2-DIMENSIONAL POROUS MEDIA MODEL

Governing Equation

Two dimensional incompressible N-S equations based on laminar assumption.

Fluid zone:
Continuum Equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

Momentum Equation:

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho_f}\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

Energy Equation:

$$\frac{\partial T_f}{\partial t} + u\frac{\partial T_f}{\partial x} + v\frac{\partial T_f}{\partial y} = a_f\left(\frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2}\right) \tag{3}$$

u, v, p and T_f are basic values for solution, which mean velocity components, pressure and temperature of the fluid.

Solid Zone
Energy Equation:

$$\rho_s C_p \frac{\partial T_t}{\partial t} = k\left(\frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2}\right) + S_t \tag{4}$$

T_t is the temperature in the solid zone, S is the source term, in this research, it can be used to demonstrate the magnetic effect.

The cooling capacity Q_{in} is evaluated as the ratio between the energy exchanged by the fluid and the cold reservoir during the hot to cold flow and the cycle time, according to the equation:

$$Q_{\dot{m}} = \frac{1}{\tau}\int_0^{\tau_1} \dot{m}(y,t)C_f(T_c - T_f(x=L,y,t))dt \tag{5}$$

The pump that provides the desired fluid mass flow rate is modeled by assigning an overall efficiency relative to an ideal thermodynamic process to these components. Using this approach, the power input to the pump is:

$$\dot{W}_{pump} = \frac{\dot{m}_f \Delta p}{\eta_{pump}\rho_f} \tag{6}$$

where η_{pump} is the overall efficiency of the pump, Δp is the pressure drop across the bed.

By performing a first law energy balance, the work input of the magnet is given as the difference between the heat rejected and the refrigeration load.

$$Q_{reject} = \int_0^{\tau_1} \dot{m}(y,t)C_f(T_f(x=0,y,t) - T_h)dt \tag{7}$$

$$Q_{refrigeration} = \int_0^{\tau_1} \dot{m}(y,t)C_f(T_c - T_f(x=L,y,t))dt \tag{8}$$

$$W_{mag} = Q_{reject} - Q_{refrigeration} \tag{9}$$

The power to run the electric motor is based on the work input to the regenerator required:

$$\dot{W}_{mag} = \frac{W_{mag}}{\eta_{motor}\tau} \tag{10}$$

η_{motor} is the overall motor efficiency.

The coefficient of performance of AMR can be calculated by:

$$COP = \frac{Q_{\dot{m}}}{\dot{W}_{mag} + \dot{W}_{pump}} \tag{11}$$

Numerical Method

The finite volume method (FVM) is used in this research, and the solid and fluid zone are discretized by using triangular unstructured mesh respectively (in Figure 1 and Table 1). In the fluid zone, the variable separation solver is adopted, in which the pressure is uncoupled with velocity by using SIMPLEC method (Semi-Implicit Pressure Linked Equation Consistent), firstly, the momentum equations are solved in the cell-based grid, then the pressure field is corrected by using pressure-correction equation, which is educed from continuum equation, and the velocity components are corrected by using pressure correction value, after that, the energy equation are solved to obtain temperature.

In the solid zone, the heat conductive equation is solved, and the source term is activated, when the magnetic field is acting:

$$S_t = \frac{\rho_s C_P \Delta T_s(H, T_c)}{\Delta t} \tag{12}$$

and boundary conditions are defined in Table 2.

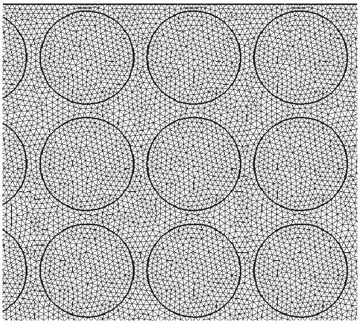

Figure 1. Schematic of Mesh

Table 1. The scheme for discretization

Pressure	Second-order central scheme
Convective term in fluid zone	Second-order upwind scheme
Diffusion term in fluid and solid zone	Second-order central scheme
Unsteady term	First-order implicit order

Table 2. The boundary conditions in this research

Inlet	Inlet velocity and static temperature
Outlet	Zero gradient
Wall between solid and fluid	Coupled wall(heat flux and temperature)
Side wall of regeneration	No-slide and adiabatic

RESULT AND DISCUSSION

In order to optimize AMR applications, we considered the same conditions that the AMR device developed in the laboratory and varied the model inputs, velocity and frequency, especially modeling the influence of velocity to the heat transfer rate from practical to fluid and pressure loss.

Figure 2 shows the temperature field for the heat transfer course under different inlet velocities. After magnetizing the fluid flow from the cold end to the hot end (in the graph is from left to right), we found that heat transfer in high inlet velocity (0.025 m/s) is already in a stable state after 0.3 second, while the heat transfer in low inlet velocity (0.005 m/s) is still in an unstable state. This condiciton indicates that high velocity can increase heat transfer.

Figure 3 shows the velocity field while the inlet velocity is 0.001 m/s. The fastest x- velocity appears in the gap between the particles and can be 10 times the inlet velocity. The y-velocity can be minus for the reverse circulation flow rate.

The velocity is changed from 0.007 m/s to 0.025 m/s and the results are shown in Figure 4 and Figure 5. Figure 4 is the heat transfer rate from particle to fluid. It is observed that a higher velocity results in a higher heat transfer rate. Figure 3 illustrates the pressure loss. The pressure loss has a positive correlation to the velocity, which is similar to the heat transfer rate. Since a higher velocity will result in a higher transfer rate as well as a higher pressure loss, an optimized velocity can be found for the best energy efficiency.

Figure 6 shows the correlation of velocity and COP, cooling capacity and Table 3 presents the reference values used in the analysis. When the frequency is 0.25 and the velocity is from 0.006 m/s to 0.014 m/s, the COP has a peak value near 6.94 for the COP and the cooling capacity with respect to the operation frequency. When the frequency is varied from 0.25 to 1.45 Hz, the COP has a peak value near 7.4. The cooling capacity increases quickly when the operation frequency is higher. A high operation frequency is required for the high cooling capacity.

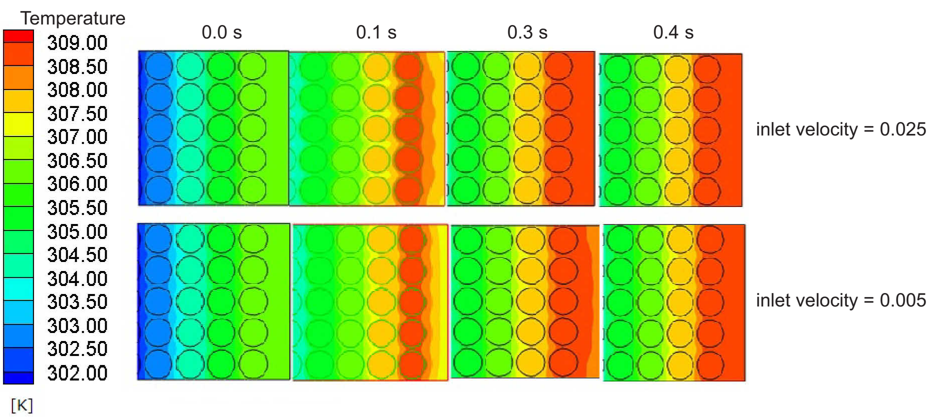

Figure 2. Temperature field for the heat transfer course under different inlet velocities.

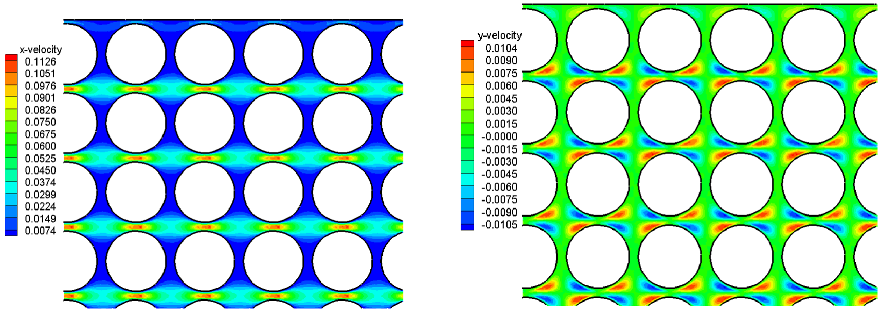

Figure 3. Velocity field. Left figure is the x-velocity field. Right figure is the y-velocity field. Inlet velocity is 0.001 m/s

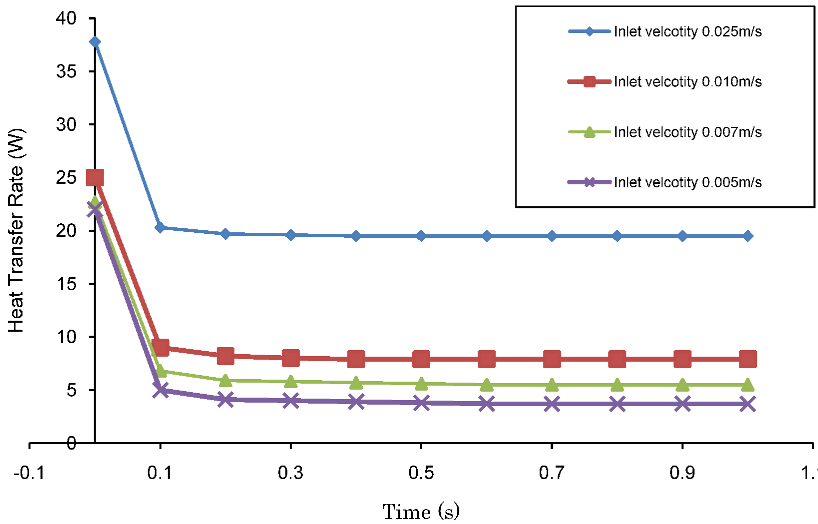

Figure 4. Heat transfer rate Vs. Time for different velocities (ΔT_{span} = 40 K)

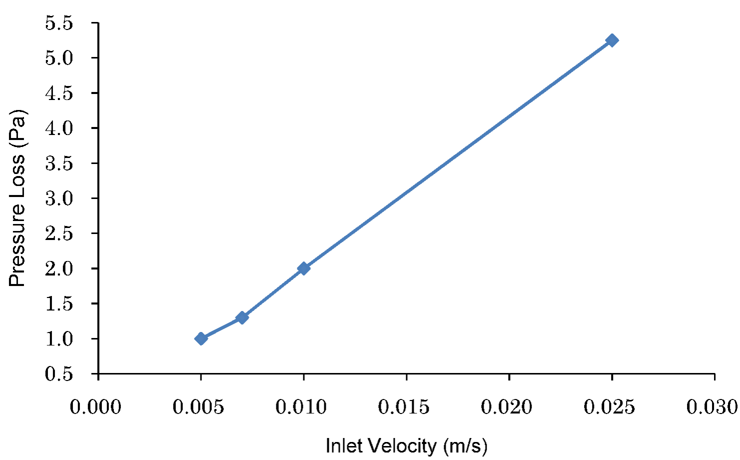

Figure 5. Pressure loss vs Inlet velocity for ΔT_{span} = 40 K

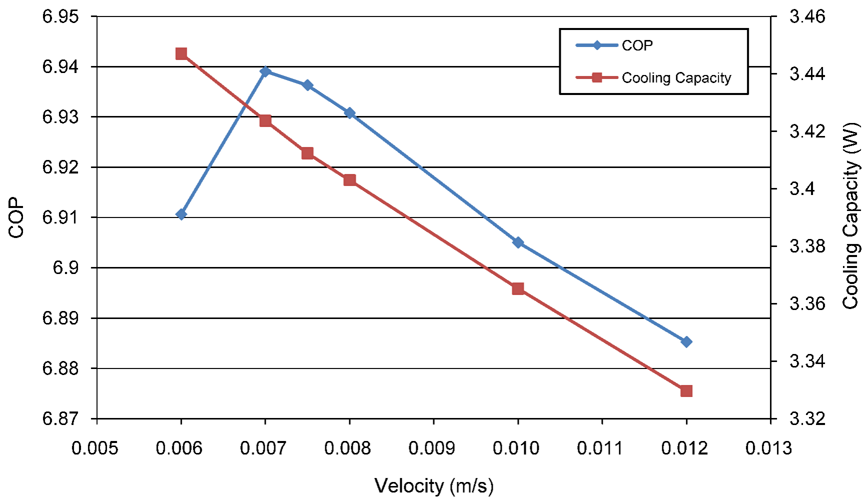

Figure 6: Velocity vs. COP and cooling capacity for $\Delta T_{span}=40K$.

Table 3. Reference values of the numerical parameters

magnetic material	Gd
Fluid	water and ethanol mixture
Total material mass	0.16kg
Regenerator size	$20.312cm^3$
Mean particle size	$500\,\mu$ m
Porosity of matrix	0.356
B_{max}	1.2T
B_{min}	0T
$\triangle T_s$	2K
T_H	313K
T_L	273K

CONCLUSION

A transient 2-dimensional porous media model of a reciprocating AMR has been constructed and solved numerically. The coupled effects of the temperature and velocity fields have been taken into account. The magneto-caloric effect is also taken into account by the inclusion of a source term in the energy equation for magnetic solid. The fluid flow through the interstitial channels formed by the magnetic particles is modeled by the two dimensional incompressible N-S equations based on laminar assumption.

In particular, detailed model results regarding heat transfer and pressure drop show that an optimized velocity can be found for the energy efficiency. The effect of changing the operating cycle frequency shows that there is a maximum COP for a given $\Delta T_{span.}$. A high operation frequency is required for a high cooling capacity.

The 2-D model will be improved to predict the performance of a layered AMR for the high efficiency of the layer regenerator bed. In addition, the 2-D model will be validated by comparison with experimental AMRs.

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